Procedure and Application for Determining the Cold Deck and Hot Deck Airflow in a Dual-Duct System

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Abstract: This paper introduces an innovative method to determine the cold and hot airflow through a dual-duct variable air volume (VAV) system. The actual building load can be identified based on the calculated airflow and temperature for both the cold and hot deck. The supply fan speed is controlled to meet the calculated load requirement. This new technology will improve the supply fan speed control and save energy. A case study was demonstrated in a commercial building during continuous commissioning practices to improve building operations and reduce energy costs. The room load, airflow rate and system resistance（S）value were recorded continuously. The results show that the supply fan operates at maximum efficiency, saving more than 90% fan power, when this new control method is used.

Key words: Dual-duct; Airflow measurement; Supply fan control

1. INTRODUCTION

The dual-duct (DD) system was popular in the 1950s and 1960s when energy was inexpensive and abundant. The DD system employs two air ducts to supply cold air and warm air to a mixing terminal unit that proportions the cold and warm air in response to a thermostat located in the conditioned space. The system is well-suited to provide temperature control for individual spaces or zones.

The dual-duct system has several advantages, e.g., it provides good air circulation and excellent humidity control. But the disadvantages, like excessive energy consumption and high initial cost, led to a decline in use after the 1973 energy crisis. However, dual-duct systems are still widely used today in offices, classrooms, laboratories, hospitals and other facilities because of their advantageous features (Joo 2004).

Continuous Commissioning (CC) has been one of the most prominent energy conservation processes for over a decade. CC has been developed to help building owners achieve energy savings, improve thermal comfort and reduce maintenance costs (Liu et al. 1999). The following three goals need to be accomplished for the DD system if possible:

1. Meet the room comfort requirement;
2. Optimize the fan speed to make the system operate at optimized system resistance; and
3. Minimize the simultaneous heating and cooling.

In a dual-duct VAV system, static pressure sensors are installed on both the hot and cold ducts. A hot deck damper is installed on the main hot deck duct. Typically, the cold deck and hot deck static pressure are reset based on the outdoor air temperature. The system will define the summer and winter mode based on the outdoor air temperature (OAT). The hot deck damper will be fully closed in summer mode. The supply fan speed is controlled to maintain the cold deck static pressure similar to the single-duct system in summer mode. The supply fan speed is adjusted to maintain at least one duct static pressure at its set point (the worst deck static pressure situation) in winter mode.

This method works well for dual-duct systems with multiple zones. However, this method can hardly ensure goals 2 and 3 at partial load conditions in winter mode. Indoor heat gain, not OAT, is the major contributor to the cooling load for large commercial buildings. For example, the room load conditions are very different when the room is full of people holding a meeting, and when the room is unoccupied, even though the OAT is at 60°F under both conditions.

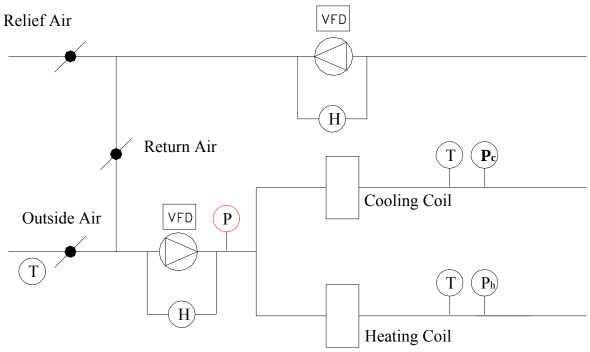

Figure 1: Typical dual-duct system

For a dual-duct system with single zone area or two zones in a large open space, the building load can be identified based on the measured deck temperature--the airflow through the cold and hot deck. The supply fan speed can be controlled to maintain the zone load requirement only. This control method can obtain our three goals for the system with one large open space, which has less load variety.

The airflow rate through the cold and hot decks can be measured by an airflow measurement station. For accuracy within 5% to 10%, the airflow measurement station requires a straight duct for 6-10 duct diameters upstream and 3 duct diameters downstream (Liu et al. 2005). However, few systems have such duct runs in the main ducts for both decks. Therefore, it is important to find an effective way to measure the airflow accurately with low installation cost. The following control sequence was developed for the VFD speed control algorithm for the supply fan of a dual-duct system.

2. METHODOLOGY

Figure 1 presents a typical dual-duct system with VFD installed on both the supply and return fans. Typically the static pressure sensors are installed on both the cold deck and hot deck. Assume that a pressure sensor is installed after the supply fan in the main duct. Then the pressure drop across the cooling coil can be obtained by

$$\Delta P_c = P - P_c \quad (1)$$

$$\Delta P_c = S_c Q_c^2 \quad (2)$$

where ΔP_c is the pressure drop across the cooling coil, S_c is the system resistance across the coil, and Q_c is the airflow through the cold deck.

For the pressure drop across the heating coil,

$$\Delta P_h = P - P_h \quad (3)$$

$$\Delta P_h = S_h Q_h^2 \quad (4)$$

where ΔP_h is the pressure drop across the heating coil and Q_h is the airflow through the hot deck.

We can easily get the pressure difference between the two pressure sensors:

$$\Delta P = \Delta P_c - \Delta P_h = (P - P_c) - (P - P_h) = P_c - P_h$$

$$= S_c Q_c^2 - S_h Q_h^2 \quad (5)$$

The total airflow measured after the supply fan is

$$Q = Q_c + Q_h \quad (6)$$

where the total airflow rate Q can be obtained by the fan airflow station (see Appendix for detail). The airflow through the cold deck Q_c and the hot deck Q_h can be solved based on Equation sets (5) and (6). There are two sets of solutions, and only one set of solutions works for the real system.

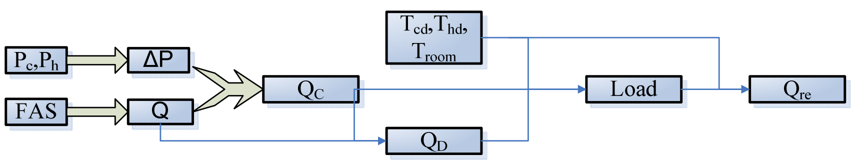

Figure 2: Flow diagram of the calculation

$$Q_c = \alpha' - \sqrt{bQ^2 + c\Delta P} \quad (7)$$

$$Q_h = \alpha - \sqrt{bQ^2 + c\Delta P} \quad (8)$$

where $a = S_c/(S_c - S_h)$, $b = = S_c * S_h/(S_c - S_h)^2$

$c = 1/(S_c - S_h)$,

$a' = -S_h/(S_c - S_h)$

We can obtain the coefficients a', b and c by measuring the total airflow, the airflow through the cold deck and static pressure at three different system resistance points. The cold deck airflow can then be identified using Equation (7). The hot deck airflow can be calculated by

$$Q_h = Q - Q_c \quad (9)$$

The load that needs to be removed can be calculated as

$$Load = [Q_C \times (T_{room} - T_{CD}) - Q_H \times (T_{HD} - T_{room})] \times \rho \times C \quad (10)$$

- If the actual load is a cooling load (Load >0), the required airflow can be calculated based on the following equation:

$$Q_{re} = \frac{Load}{(T_{room} - T_{CD})} \quad (11)$$

- If the actual load is a heating load (Load<0), the required airflow can be calculated based on the following equation:

$$Q_{re} = \frac{Load}{(T_{HD} - T_{require})} \quad (12)$$

Then the supply fan speed is directly controlled to meet the airflow requirement CFM_{re}. *Note:* if the calculated fan speed is lower than the minimum fan speed, keep the fan at minimum fan speed.

Based on the above procedure, Figure 2 shows the flow diagram of the calculation.

Please Note: this control sequence was specifically developed only for large open spaces with less load diversity. The sequence is not appropriate for a typical office space that may have very different load conditions in individual rooms.

3. CASE STUDY

The case study building, located in Omaha, Nebraska, was originally built in 1960. This office building includes a 16-story tower and a two-story addition to the east.

The Cloud Room, which is served by AHU 16 with a conditioned area of 5000 ft^2 ($465m^2$), has a total of 12 terminal boxes and four thermostats on the 16^{th} floor. The Cloud Room is used for holding meetings or parties. It is a dual-duct system with aconstant volume pneumatic-control box. Figure 3 shows the floor and terminal box layout plan.

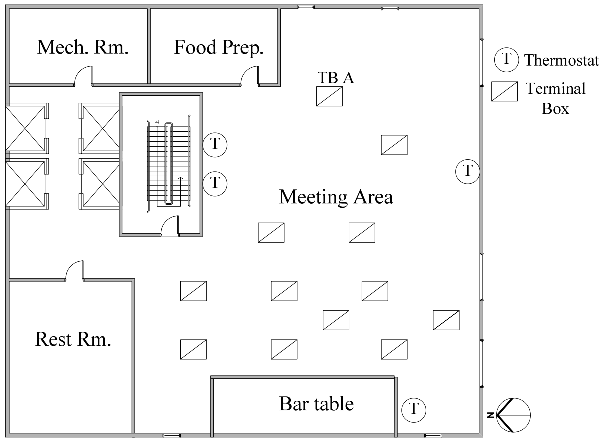

Figure 3: the Sixteenth floor layout plan

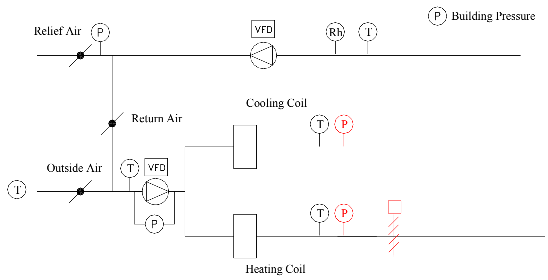

Figure 4: Schematic diagram for AHU16

The cold deck and hot deck damper at the terminal box are interlocked using one actuator. The supply fan is 15 HP and the return fan is 5 HP without variable frequency drive (VFD) installed. The building pressure set point is 0.1 in. w.g. (25 Pa) which is controlled by relief air damper. The design airflow is 9760 CFM (4606 m^3/s) and design fan head is 4.6 in. w.g. (1146 Pa).

The system retrofit includes adding a control damper on the main hot deck and installing a VFD on both the supply and return fans. The fan power reading from the VFD was sent to Energy Management Control Systems (EMCS) as an analog signal. A fan head-based fan airflow station (FAS) is installed to measure the total airflow rate. The schematic diagram of AHU16's existing and added sensors is shown in Figure 4.

The new control sequences are listed as follows:

1. *Supply fan speed control*

After the CC implementation, the supply fan speed was controlled following the rules below:

[1] Summer mode (OAT>60ºF)

When the outside air temperature is higher than 60°F, the system will be switched to a single-duct system by totally shutting off the hot deck damper. The supply fan speed will be controlled to maintain the constant system resistance. The system resistance can be calculated based on the fan head and air flow measured by FAS.

The thermostat can be adjusted to full cooling to make all the terminal boxes open when the supply was operating at full speed. Then the total airflow rate and fan head was measured at summer mode. The measured airflow is 8400 CFM and measured fan head is 4.8”. Assume the dimensionless S value (H/Q^2) equal to 1 at design condition, the actual system resistance can be calculated as

(4.8”/4.6”)/(8400CFM/9760CFM)^2=1.41

The actual system resistance is a little higher than the design condition because the system installation and the aging terminal boxes. Then the supply fan speed is controlled to maintain the system resistance at 1.5.

The airflow rate Q is obtained by the FAS using Equation (13).

$$Q' = \frac{-a_1\overline{\varpi} - \sqrt{a_1^2\overline{\varpi}^2 - 4a_2(a_0\overline{\varpi}^2 - H)}}{2a_2} \quad (13)$$

where ϖ is the fan speed ratio, and H is the measured fan head. The identified in-situ fan curve (dimensionless) coefficients are

a_0=1.004, a_1=0.8622, a_2=-1.1325

[2] Winter mode (OAT<60ºF)

When the outside air temperature is less than 60°F, there are two conditions for control:

- If the actual load is a cooling load, then the required airflow can be calculated based on Equation (11).
- If the actual load is a heating load, then the required airflow can be calculated based on Equation (12).

If the calculated fan speed is lower than the minimum fan speed, maintain the minimum fan speed at 15% (around 950 cfm). The load can be calculated following the procedure shown in Figure 2.

We measured the airflow rate and pressure difference at three system resistance points. Table 1 shows the measurement results.

Table 1: Field measurement results

SF speed (%)	CFM (cold deck)	CFM (hot deck)	Total CFM
60.3%	2754	2023	4777
60.3%	2451	2756	5208
100.0%	3878	4525	8403

Based on our measurements, we can obtain the coefficients for Equation (7):

$$Q_c = 1.2 * Q - \sqrt{0.326 * Q^2 + 7867875 * \Delta P}\ .$$

The equation was verified using the above equation at five different system resistances. The differences between the direct measured airflow and the calculated airflow are within 5%.

[3] Hot air damper control:

The hot deck damper will be fully open when the OAT is lower than 60ºF, and it will be fully closed when the OAT is higher than 60ºF.

2. *Return fan control:*

Modulate the return fan speed to maintain return airflow rate set points based on the airflow station when the supply and return fans are both on.

The return air fan is controlled to maintain the calculated return airflow rate at a set point of 800 CFM (378m^3/s) less than the supply airflow rate. The set point was determined by an experiment in which the value from the building pressure sensor was maintained at a constant positive value of 0.1 in. w.g. (25 Pa)on a very calm day.

When the supply fan speed is lower than 20%, the return fan will be shut off.

3. *Temperature reset*

- Before CC :

Figure 5 shows the temperature reset schedule for the cold and hot deck

- After CC:

Reset the hot deck and cold deck set point based on the outside air temperature and return air relative humidity.

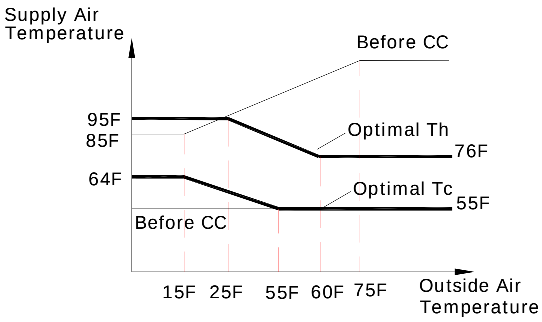

Figure 5: Supply air temperature reset schedule

[1]. Set the hot deck set point as in Figure 5.

[2]. Set the cold deck set point at 55ºF if the return air humidity is higher than the set point (60%). Otherwise, reset the set point based on the set value in figure 5.

4.RESULTS

The new control sequence was implemented in March 2006. The temperature and static pressure sensors were calibrated and the measurement information was trended using EMCS. The outdoor air temperature, return air temperature, deck supply air temperature, fan speed, airflow for both decks, fan head, fan power and building pressure were automatically recorded every 15 minutes. A total of 2961 sets of data was collected from 4/13/2006 to 5/15/2006.

The system control will be evaluated by the three goals we mentioned in the introduction section:

I. Meet the room comfort requirement.

Figure 6 shows the fan speed control, outdoor air temperature and return air temperature from 4/13 to 5/13. The figures shows that the supply fan speed can be controlled well using the load calculation method, and the room temperature can be maintained at proper range.

Figure 7 shows the building pressure control results using the fan airflow station. The high building pressure readings were resulted from the wind gust weather patterns in April and May. For example, the average wind gust speed was 17.8m/s (**39.8 mile/hr**) during the day time, according to the National Weather Station database. Even though the differences between the supply airflow rate and the return airflow rate were maintained at 800 ft^3/min, the values from the building pressure sensor were agitated. The accuracy of the building pressure sensor was significantly impaired by wind. The building pressure was controlled between ranges of ranges of 0 to 0.10 in. w.g. for more than 75% of the time. Figure 7 proves that the FAS works better than building pressure control by actual building pressure measurement.

II. Optimize the fan speed to make the system operate at optimized system resistance.

Figure 8 presents the supply fan speed versus the outdoor air temperature. The figure shows that the fan speed is controlled below 30% when the outdoor air temperature is lower than 60ºF. The supply fan speed was reduced significantly using the load identification control.

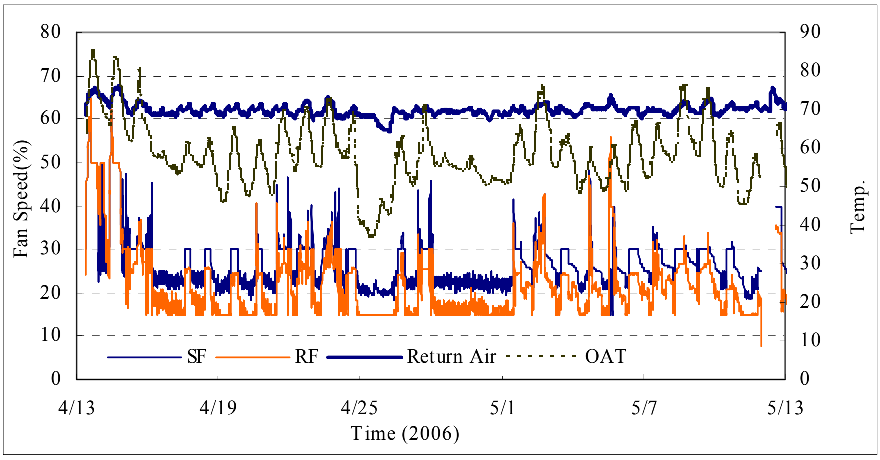

Figure 6: Measurement results (2006)

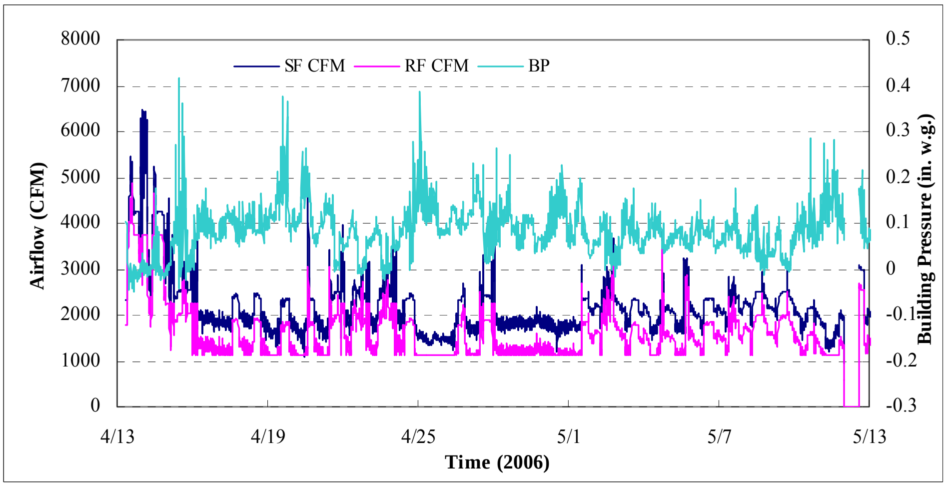

Figure 7: Building pressure trended from EMCS (2006)

Figure 9 shows all the fan operation points (fan head and airflow rate) in the dimensionless manufacturer's fan performance curve chart. The x axis and y axis are calculated based on the design airflow and design fan head. S is the system resistance, calculated by dividing fan head by the square of the airflow rate. The highest efficiency is 70% when the S value is equal to 1. The fan efficiency is about 66% when the system resistance S=2. This figure shows that most of the fan operation points are within the best efficiency range (between 66% to 70%). Figure 10 shows the supply fan power consumption before and after CC implementation. It shows that the new control sequence can save more than 90% fan power when OAT<60˚F using this new control method.

III. Minimize the simultaneous heating and cooling.

Using the new supply fan control algorithms and load identification control sequence we developed, the simultaneous heating and cooling is minimized.

5. CONCLUSIONS

This paper introduces a new method to determine the cold and hot airflow through a dual-duct VAV system and its application. The actual building load can be identified based on the calculated airflow and temperature for both the cold and hot deck. This will significantly improve the supply fan speed control and save energy.

The supply fan airflow is measured by FAS. The cold deck airflow is calculated using supply airflow, and the static pressure difference of the cold and hot decks. The supply fan was controlled to meet the load requirement. The other new technologies include building pressure control and duct static pressure reset based on the measured airflow. The results show that this method can meet the room comfort requirement and optimize the fan speed to make the system operate at optimized system resistance. It can also minimize the simultaneous heating and cooling and significantly reduces energy costs. The case study also proves that the fan airflow station is a reliable and accurate solution for building pressure and airflow control in VAV systems.

The control sequence was specifically developed only for large open spaces that have less load diversity. The sequence is not suitable for a typical office space that may have very different load conditions in individual rooms. The application of the cold deck and hot deck airflow measurement method to a dual-duct multi-zone system needs further study.

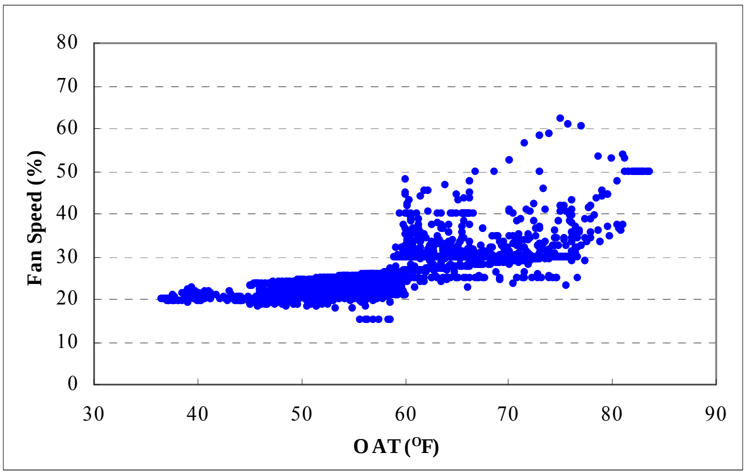

Figure 8: Supply fan speed vs. OAT (2006)

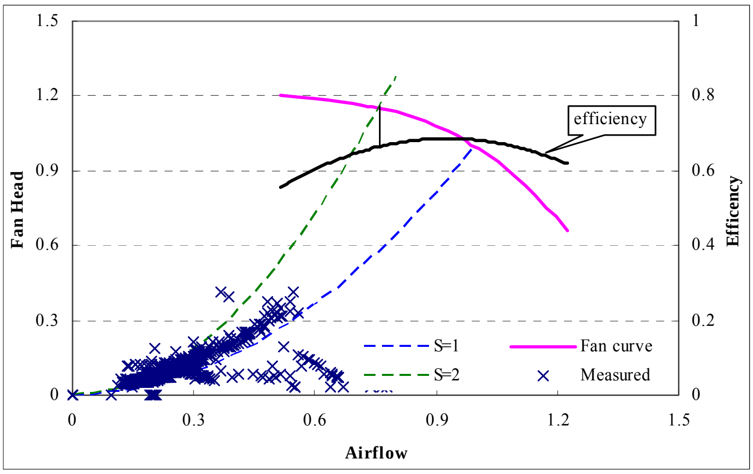

Figure 9: Supply fan speed control and fan efficiency

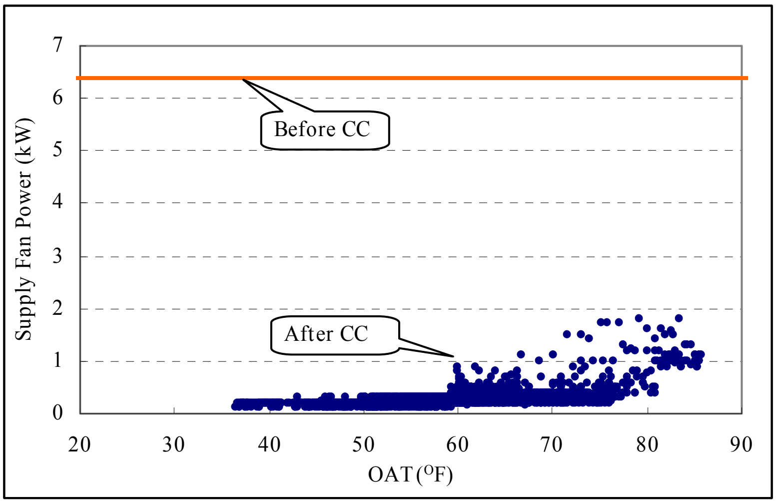

Figure 10: Supply fan power consumption

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REFERENCES

[1]. Joo, I. 2003, Optimal Design and Control in Dual-duct Air-handling Units. Doctoral Dissertation, University of Nebraska- Lincoln.

[2]. Liu M. 2002. Variable Speed Drive Volumetric Tracking (VSDVT) for Airflow Control in Variable Air Volume (VAV) Systems. *Journal of Solar Energy Engineering*.

[3]. Liu, M., Liu, G., Joo, I., Song, L. and G. Wang. 2005. Development of In Situ Fan Curve Measurement for VAV AHU Systems. *Journal of Solar Energy Engineering*, 127(2):287-293.

[4]. Liu, G., Joo, I., Song, L., and M. Liu. 2003. Development of In-situ Fan Curve Measurement with One Airflow Measurement. *Proceedings of the International Conference for Enhanced Building Operations (ICEBO)*, October 13-14, Berkeley, California.

[5]. Liu, M., Claridge, D.E., Haberl, J.S., and W.D. Turner. 1999. Improving Building Energy Systems Performance by Continuous Commissioning. *Energy Engineering,* May.

APPENDIX

The fan airflow station was developed by Liu in 1997. It determines the airflow using fan law based on measured fan speed, fan head, and the pre-determined relationship of the fan head and fan airflow (fan curve) under a given fan speed. Under full speed, the fan curve equation can be represented as a 2nd order polynomial equation:

$$H = a_0 + a_1 \cdot Q + a_2 \cdot Q^2 \qquad (1)$$

where H is a fan head [inch H_2O, Pa] under full fan speed.

Q is an airflow rate [ft^3/min, m^3/s] under full fan speed.

a_0 , a_1 *and* a_2 are fan curve coefficients.

Equation (1) works well at the normal operation region for most fans in AHUs. If the fan runs under partial speed by the VFD, the fan curve can be represented as Equation 2, which is derived by the fan laws:

$$H = a_0 \overline{\omega}^2 + a_1 \cdot Q' \overline{\omega} + a_2 \cdot Q'^2 \quad (2)$$

where: ϖ is a current fan speed over 100% fan speed.

Q' is a calculated airflow rate under partial speed of the fan.

The airflow rate at any measured fan head and fan speed can be obtained by solving Equation (2). The actual airflow rate is calculated by Equation (3).

$$Q' = \frac{-a_1 \overline{\varpi} - \sqrt{a_1^2 \overline{\varpi}^2 - 4a_2 (a_0 \overline{\varpi}^2 - H)}}{2a_2} \quad (3)$$

The fan airflow station includes a differential pressure transducer and fan speed transducer. The fan speed transducer may be replaced by the control system command to the VFD. The fan airflow station can be implemented using a typical Energy Management and Control System (EMCS). Figure 1 shows a schematic diagram of the airflow measurement and control in a typical VAV system using fan airflow stations. The return airflow set point is determined as the difference of the supply airflow and the sum of the building exhaust and air exfiltration. The return airflow is determined using the measured return fan head and speed. The return fan speed is modulated to maintain the return airflow set point. The theoretical model of the fan airflow station has been experimentally tested, and it was found that the model closely agreed with the experimental values. This method is a very effective way to measure the airflow accurately under all weather conditions. The fan airflow station was also implemented by ESL in a number of AHUs for building pressure control.

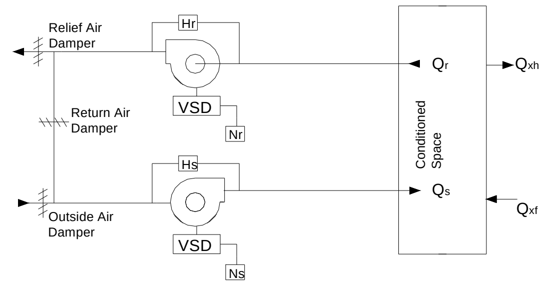

Figure 11: Schematic of airflow in a VAV system using fan airflow station